TRANSFER OF HEAT FROM HELIUM AT HIGH TEMPERATURES-- PHYSICAL CHARACTERISTICS OF CERAMIC HEAT EXCHANGERS

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Preliminary cost studies indicate that it should be economically possible to use nuclear energy as process heat for certain endothermic reactions. The Federal Bureau of Mines' interest at present is utilizing heat from the fission of nuclear fuels to gasify coals with steam to produce synthesis gas. Of primary concern in the study of the use of nuclear heat for gasification of coal and for other industrial purposes is the problem of effective transfer of heat from the gas-cooled nuclear reactor-core to a reaction chamber where the endothermic reaction takes place. In order to remove the heat from the coolant-gas, it is necessary to design a heat exchanger capable of performance at temperatures much in excess of those ordinarily encountered in heat exchange equipment. Heat exchangers constructed of available metals will not withstand prolonged use at high temperatures. This suggests the investigation of certain ceramic materials capable of withstanding high temperatures and other severe conditions of corrosion and erosion.

Much data are available in the literature on heat transfer by forced convection to and from gases. Summaries of previous work have been published by McAdams (15) and others (8, 11). Most of the existing experimental data, however, refer primarily to heat exchange in metal tubes, and do not extend into the range of higher temperatures in which interest has increased in many current engineering applications.

An experimental investigation was undertaken at the Morgantown Coal Research Center of the Bureau of Mines, U. S. Department of the Interior, Morgantown, W. Va., to obtain heat-transfer information over a wide range of surface and fluid temperatures, with helium as the heat-transfer medium. As part of this broad program, an investigation was made of the transfer of heat from helium (flowing through smooth tubes) to the tubular surfaces of a vitreous alumina heat exchanger. The effects of such variables as tube-wall temperature and inlet helium temperature were investigated. The feasibility of the use of ceramic materials in a heat exchanger and some of the factors involved in the design and fabrication of a ceramic exchanger were also studied.

In most of the forms of heat-exchange apparatus based on forced convection the velocity of the fluid is maintained at a sufficiently high level to assure a turbulent flow. In this investigation, however, it was not possible to attain turbulent flow because of the low density of helium at atmospheric pressure and the increasing viscosities of gases with temperature. The pressure could not be increased above atmospheric as it was necessary to hold the helium loss from the system within practical limits. Consequently, this investigation was conducted in the lower laminar flow region at atmospheric pressure.

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Complete mathematical solution is known for only a relatively few cases of heat transfer. Since this problem involves so many variables, a mathematical expression for the transfer of heat through the fluid film is usually developed by the method of dimensional analysis. Attempts to solve the Fourier-Poisson equation have met with little success, mainly because of the hydrodynamic problem that must be solved simultaneously. The completed solutions of the Fourier-Poisson equation depend on convenient assumptions concerning the motion of the fluid.

Drew (6) eliminates most of the proposed solutions and lists only three for the case of a jacketed tube or pipe, i.e. those of Graetz, Leveque and Russell. Several investigators (4, 7, 16) evaluated Graetz' results for the conduction of heat in fluids moving in laminar flow. Other equations (5, 13, 14, 18) derived for the calculation of the Nusselt number from the velocity distribution vary in form due to the various assumptions used in deriving them.

For laminar flow, McAdams (15, 19) recommends an empirical expression for calculating the inside-film heat-transfer coefficient:

$$\left(\frac{h}{CV\rho}\right)\left(\frac{C\mu}{k}\right)^{2/3}\left(\frac{\mu_w}{\mu}\right)^{0.14} = 1.86\left(\frac{D}{L}\right)^{1/3}\left(\frac{DV\rho}{\mu}\right)^{-2/3} \quad (1)$$

This relationship applies for Reynolds numbers below 2100, but it would give incorrect heat-transfer coefficients for very low "Re" numbers.

Theoretical formulas, such as those of Graetz, for heat transfer to fluids in viscous flow in tubes, do not take into consideration the effects of the radial temperature gradient on the axial and radial components of velocity. Because of the temperature coefficient of viscosity, a radial viscosity gradient results that affects the distribution of velocities considerably compared to that prevailing in isothermal flow. This presents a problem of deciding where, and at what temperature, the physical properties of the fluid should be evaluated. Colburn (4) obtained good correlation of data on heat transfer by employing film properties, but had to supplement his correlation by introducing a viscosity correction ratio.

Sieder and Tate (19) simplified calculations by basing their correlations on main stream properties. To express the interaction of the viscosity gradient on the velocity distribution, they included in the usual correlation of dimensionless numbers an additional dimensionless group, μ_a/μ_w, i.e., the ratio of the viscosity at the average helium temperature to that at the wall temperature.

The correlations heretofore discussed were made at relatively low temperatures; only limited data exists for heat transfer at higher temperatures. Zellnik and Churchill (20) obtained data for the transfer of heat from air in turbulent flow inside a tube for temperatures of 480 - 2000°F. and flow rates corresponding to Reynolds numbers from 4500 to 22,500. Others (10) have published some data on heating air inside a tube with wall temperatures up to 2950°F. More recently, Ramey, Henderson and Smith (17) determined heat-transfer coefficients for air flowing in a 2-inch pipe at temperatures of 500 - 1200°F., with gas-to-wall temperature differences of 300 to 1000°F., and Reynolds numbers from 2000 to 20,000. Their results include data on steam over a range of Reynolds numbers from 5000 to 60,000.

However, all these high-temperature heat-transfer studies pertain to turbulent flow. The data pertaining to turbulent flow at high temperatures deviated from those relating to low-temperatures when correlated by the proposed theoretical equations.

Apparatus and Procedure

A schematic diagram of the apparatus used for our study of heat transfer at high temperatures, is shown in Figure 1. Helium was recycled in a closed circuit by means of a rotary positive-displacement blower. The helium from the pump was passed through a surge tank and heated to the desired temperature in a resistance-type electric furnace. This temperature was measured by a shielded thermocouple placed in a mixing chamber located between the outlet of the furnace and the inlet to the "test-section", i.e., that part of the train in which the tested heat exchanger is inserted. The helium temperature was measured also at the outlet from the test-section before entering the cooler. The helium cooled to room temperature was passed through two parallel-connected calibrated rotameters to measure the rate of flow of the heat transfer medium through the closed circuit. The test-section, shielded thermocouples, and adjoining tubing were thermally insulated. Additional helium to replace that which leaked from the system was supplied through a pressure-regulator. The flow rate of this "make-up" gas was indicated by a calibrated rotameter.

Compressed air was supplied as heat receiving medium to the annulus of the test-exchanger. The wall temperature of the inner tube of the test-exchanger was regulated by changing the rate of flow of the air through the annular space. High tube-wall temperatures were obtained by preheating this air in a commercial electric resistance furnace. The direction of flow of gases in the exchanger could be altered from concurrent to countercurrent, or vice-versa, simply by switching the inlet and outlet valves for the air or any other cooling fluid used as heat absorbers. The temperature of the cooling air was measured at the inlet and outlet of the heat exchanger; its flow-rate was measured by a calibrated rotameter.

A test-exchanger made of vitreous alumina, with a length-to-diameter ratio of 60, was used (Figure 2). The vitreous alumina inner tubing was surrounded by a larger diameter tube of the same material, and the cooling air was passed through the annular jacket thus formed. Tube wall temperatures were measured by means of (Pt)-(Pt+10 percent Rh) thermocouples located at ten different points along the tube length. Each thermocouple was placed in a slot cut halfway through the tube wall and covered over with alumina cement. The thermocouples were connected by means of a selector switch to a potentiometer.

The physical properties of the helium were computed by extrapolation of available published data. The thermal conductivity and viscosity data were taken from the results of work done at the National Bureau of Standards by Hilsenrath and Touloukian (9). The specific heat of helium at constant pressure (1) was taken as 1.24 B.t.u./(lb.)(°F.). This value agreed with that calculated from the Prandtl numbers and viscosity and thermal conductivity data of Hilsenrath and Touloukian and was assumed constant, since the specific heat of helium does not vary with temperature.

Experimental Results and their Evaluation

The average temperature of the helium ("hot gas"), T_a, was taken as the average of the inlet, T_i, and outlet, T_o, temperatures. Each of these temperatures was measured by a "shielded thermocouple" arrangement through which the hot helium had passed and was thoroughly mixed.

Average inside tube-wall temperatures were determined for various flow rates of helium and cooling air; typical results are illustrated for counterflow of gases in Figure 3, and for parallel flow in Figure 4. The rate of heat transfer (Q), the helium flow rate (V), average wall temperature (t_w), and the helium temperature at the inlet (T_i) are shown as parameters. The curves obtained are typical temperature distribution curves; similar trends were obtained for other conditions of flow and gas temperatures.

The average wall temperatures of the inner tube of the test-exchanger were obtained by measuring the area under curves similar to those in Figures 3 and 4 and dividing it by the length of the heat-transfer area. Thermocouples embedded in the tube wall were assumed to measure the temperature of the inside surface of the tube. The fact that the calculated temperature gradients through the wall were found to be negligible, less than 3°F., justifies the assumption. The average inside-film heat-transfer coefficient (h_i) was obtained from the experimental data by the relation:

$$h_i = \frac{WC\ (T_i - T_o)}{A\ (t_w - T_a)} \qquad (2)$$

As pointed out previously, the usual method of correlating experimental data on heat transfer by forced convection is being employed by determining the effects of significant variables on the rate of heat transfer and combining these variables into an empirical equation by means of dimensional analysis. The equation obtained by this method is:

$$\frac{h_i D}{k} = K \left(\frac{DV\rho}{\mu}\right)^m \left(\frac{C\mu}{k}\right)^n \left(\frac{L}{D}\right)^p \qquad (3)$$

where K is a constant and m, n and p are exponential constants. McAdams (15) suggested a simpler form of this equation, derived by assuming that m, n and p are equal:

$$\frac{h_i D}{k} = K \left(\frac{4}{\pi}\ \frac{WC}{kL}\right)^a \qquad (4)$$

The experimental data obtained in this investigation were correlated by use of both types of equations. However, Equation (3) gave a better representation of the data from our ceramic heat exchanger for the L/D ratios employed. To establish a more general equation valid for any L/D ratio, additional investigation is necessary to cover a wider range of experimental conditions.

The experimental data obtained were first correlated in terms of Equation (4). For this purpose, the viscosity and thermal conductivity data used had been computed for the average gas temperature.

Data obtained by use of helium cooled by air in a vitreous-alumina heat exchanger, having a length-diameter ratio of 60, are presented in Figure 5. The data shown refer to inlet-gas temperatures of 1450 - 2665°F. and Graetz numbers of 0.03 - 5. A group of three parallel straight lines, each for a different condition of cooling-air flow, has been obtained. The plot shows conclusively that either by reducing the velocity of the cooling air or by heating it, the trend -lines shifted in position, and a different value was obtained for the constant, K. That is to say, there is a difference in Nusselt number for any selected value of Graetz number. Since varying the velocity or increasing the inlet temperature of cooling air had the effect of varying the tube-wall temperature, it seemed obvious that a satisfactory correlation would require inclusion of the surface temperature, or of properties that are dependent upon the latter.

When the average temperature is used for the evaluation of heat transfer properties, no allowance is being made for the variation of these properties over the cross section of the flowing fluid. However, these variations influence the heat exchange within the fluid. The Seider and Tate viscosity correction takes into account variations of viscosity, but ignores the other properties. The use of a gas-to-wall temperature ratio, on the other hand, corrects for variations in the velocity profile with temperature. But, as these velocity variations are caused by changing physical properties due to temperature changes, the T_a/t_w ratio actually corrects for all changes in the physical properties with temperature. Therefore, application of the temperature-ratio corrections would be expected to result in a better correlation. Consequently, the $\frac{4}{\pi}\frac{WC}{kL}$ group was supplemented by addition of the T_a/t_w ratio. The final form of Equation (4) thus became:

$$\frac{h_iD}{K} = K\left(\frac{4}{\pi}\frac{WC}{kL}\right)^a \left(\frac{T_a}{t_w}\right)^b \tag{5}$$

where: h_iD/k is the Nusselt number (Nu), and WC/kL is the Graetz number (Gz).

Several runs were made holding the Graetz number constant in order to determine the extent of the dependence of Nusselt number on the temperature ratio. The results of these tests are shown in Figure 6. The average value of the exponent b, obtained by taking the slope of these curves, was found to be -0.9 for values of T_a/t_w less than approximately 3. It can be seen from Figure 6 that any further increase in the gas-to-surface temperature ratio above 3 has practically no effect on the Nusselt number.

In laminar flow, when a fluid flows isothermally the velocity profile is assumed to be parabolic, with a maximum velocity at the center axis of the pipe or tube, and zero velocity at the wall. The flow may be thought of as concentric cylindrical elements of fluid moving relative to each other with little or no mixing of the layers. If the hot stream of gas is being cooled, as in this investigation, the viscosity of the gas near the wall is lower than that of the main stream of the fluid; consequently, the fluid near the wall travels at a higher velocity than normally when the wall is not cooled. For this to happen, some of the gas from the center of the tube must flow toward the wall to maintain the increased velocity at the tube-wall. Thus, the cooling of the fluid causes a radial component of velocity that modifies the nature of the laminar flow.

Since in cooling the gas the parabolic distribution of velocity is distorted, correction should be applied. As the difference in gas and wall temperatures increases, T_a/t_w increases, it is expected that the velocity profile would tend to approximate that of a well-mixed fluid, and the properties would not vary appreciably across a section of the tube. Further increase in T_a/t_w would not be expected to change the velocity profile appreciably; the properties of the fluid would remain essentially the same across the tube section and no correction would be needed.

For values of T_a/t_w less than 3, Equation (5) can be written as:

$$\frac{h_iD}{k} = K\left(\frac{4}{\pi}\frac{WC}{kL}\right)^a \left(\frac{T_a}{t_w}\right)^{-0.9} \tag{6}$$

This equation was rearranged into a more useful form for plotting:

$$\log\left[\frac{h_iD}{k}\left(\frac{T_a}{t_w}\right)^{0.9}\right] = a\log\left(\frac{4}{\pi}\frac{WC}{kL}\right) + \log K \tag{7}$$

The value of constant "a" was found to be 0.86 from a plot of this equation, Figure 7, by measuring the slope of the trend-curve (a straight line) obtained by the best free fit. From the same plot, for the value of "K" 1.10 was obtained as the intercept on the ordinate at unity on the abscissa. The plotted data have been corrected for differences between the gas and wall temperatures.

Substituting the values of K, a and b into Equation (5), the following equation:

$$\frac{h_iD}{k} = 1.10\left(\frac{4}{\pi}\frac{WC}{kL}\right)^{0.86}\left(\frac{T_a}{t_w}\right)^{-0.9} \tag{8}$$

was obtained for values of T_a/t_w less than 3.

For values of T_a/t_w greater than 3, a plot of the experimental data based on Equation (4) gave a similar straight line (Figure 8). No correction has been applied in this plot for differences between the gas and wall temperatures for the reasons already discussed.

A better correlation of the experimental data was obtained by Equation (3) which contains the dimensionless groups associated with turbulent flow:

$$\frac{h_i D}{k} = K \left(\frac{DV\rho}{\mu}\right)^m \left(\frac{C\mu}{k}\right)^n \left(\frac{L}{D}\right)^p \qquad (3)$$

where $h_i D/k$ is the Nusselt number (Nu), $DV\rho/\mu$ is the Reynolds number (Re), and $C\mu/k$ is the Prandtl number (Pr). However, the number of length-diameter ratios (L/D) investigated was insufficient to determine their effect on the Nusselt number; therefore, the correlations presented here are applicable only to the values of L/D investigated.

In Figure 9 the heat-transfer coefficients are correlated with other pertinent variables in terms of the Reynolds (Re), Prandtl (Pr), and Nusselt (Nu) numbers. Namely, the ratio of the Nusselt number to the cube root of the Prandtl number ($Nu\ Pr^{-1/3}$) was plotted against the Reynolds number (Re). Two parallel straight lines, one for a lower and one for a higher flow-rate of cooling-air, have been obtained. For computing the numerical values of these dimensionless groups for plotting, the values of viscosity, thermal conductivity, specific heat and other physical properties of the helium have been estimated at the average stream temperatures.

If corrections are incorporated for differences in helium-stream and wall temperatures, the curve in Figure 10 is obtained for values of T_a/t_w less than 3. The slope of this trend line is 0.86 and the intercept on the ordinate at Re = 1 is 0.028. Thus, Figure 10 represents a plot of the following equation:

$$\frac{h_i D}{k} \left(\frac{T_a}{t_w}\right)^{0.9} \left(\frac{C\mu}{k}\right)^{-1/3} = 0.028 \left(\frac{DV\rho}{\mu}\right)^{0.86} \qquad (9)$$

Transposing, Equation (10) is obtained, which adequately represents the experimental data:

$$\frac{h_i D}{k} = 0.028 \left(\frac{DV\rho}{\mu}\right)^{0.86} \left(\frac{C\mu}{k}\right)^{1/3} \left(\frac{T_a}{t_w}\right)^{-0.9} \qquad (10)$$

for values of T_a/t_w less than 3.

The heat-transfer data obtained in this work is of limited scope due to the low helium flow-rates used. The heat-transfer curves should not be extrapolated beyond the range of values plotted, as a change in the slope of the curve for Graetz numbers above 5 has been indicated in our investigations.

The gas-to-wall temperature ratio was used as a correction factor instead of the viscosity ratio correction of Sieder and Tate (19). Consequently, the gas-wall temperature correction would not be applicable to a gas whose physical properties are affected by temperature differently than those of helium. Likewise, the point at which the temperature ratio ceases to influence the heat-transfer curve would be expected to be different for other gases.

Free convection effects were neglected in this investigation since the tubes used were horizontal and were of relatively small diameters. In larger tubes, free convection would be expected to influence the heat transfer as long as laminar flow was maintained.

In view of the low flow-rates, it was assumed that end effects were negligible and that laminar flow was maintained within the test section.

Problems in the Design of Ceramic Heat Exchangers

The difficulties encountered in the design and construction of our experimental vitreous alumina exchanger indicate that several construction problems remain to be solved before a ceramic heat exchanger could be produced commercially. The greatest of these problems is that of making and maintaining several gastight seals, some of which must withstand temperatures exceeding 2500°F.

These seals must be made with some type of cementing material, which when dried and fired will have a coefficient of expansion similar to that of the materials joined. As much of the piping and auxiliary apparatus used with the exchanger is metallic, it is necessary to join materials with greatly different coefficients of expansion. No satisfactory solution to this problem has yet been found.

In our experimental work the most effective joints, both ceramic-to-ceramic and ceramic-to-metal, were obtained by using a mixture of 60-70% fine grain (ball-milled for 16 hours) pure alumina plus 30-40% commercial sodium silicate solution. After the cement had air-dried, the joints were fired at temperatures above 2200°F. Several successive coatings of this cement were applied, and the drying-firing process was repeated. However, joints of this type have not always been satisfactory.

After approximately 1000 hours of operation, our experimental alumina heat exchanger was removed from the test circuit to test the seals for gas leaks. While some of the joints were found to be gastight at low pressures (5 p.s.i.g.) others were not. Also, after an extended period of operation at high temperatures, the alumina cement had a tendency to peel off the metal in case of metal-to-ceramic joints.

The difficulty of constructing effective ceramic seals was also realized under actual operating conditions. In all tests enough helium was supplied to the cycling system to maintain a slightly positive pressure in the test loop. With higher pressure drops, caused by attempts to increase the helium flow rates in the system, it became necessary to increase the helium makeup to maintain the pressure. The makeup rate (helium loss) eventually became so great that it was no longer practical to continue to increase the gas cycling rate. Thus, pressure drops above 16-inch w.g. could not be tolerated. At very low flow-rates, on the other hand, experiments have been continued for several days without adding any helium as makeup.

In addition, definite precautions had to be taken to prevent failure of the tubes from thermal shocks. Although, vitreous alumina has remarkably high resistance to thermal shocks compared to most ceramic materials, elaborate precautions had to be taken to avoid fracture from thermal stresses. Care was taken during heating and cooling periods to avoid large thermal gradients likely to cause tube failure.

Efforts have been made recently by several investigators (1) to predict the ability of basic components of ceramic materials to withstand thermal stresses encountered in service. Of particular interest in the investigations of Baroody and associates (2) was the high resistance to thermal fracture exhibited by thin-walled tubes such as might be used in ceramic heat exchangers.

Another problem in the design of ceramic exchangers is that of expansion. Any type of expansion joint adds to the task of maintaining a leak-proof cycling system. When providing for expansion, it should be desirable to keep the components of the exchanger in a state of compression rather than tension, as ceramics are relatively weak in tension. No special precautions were necessary for the expansion of the exchanger parts in this investigation since the outer tube was large compared to the inner tube, and the stresses encountered were considered safely below the limit.

Conclusions

Correlations for the prediction of the average inside-film coefficients of heat-transfer for the removal of heat from gases in the lower region of laminar flow, with high temperature differences prevailing, must include a correction for the variation of fluid properties up to a certain value of gas-to-wall temperature ratio. Beyond this point, any increase in this temperature ratio ceases to influence the heat transfer relationship.

Within the limits of this investigation, and for gas-to-surface temperature ratios less than 3, the experimental data for helium may be represented by the equation:

$$Nu = 0.028\ (Re)^{0.86}\ (Pr)^{1/3}\left(\frac{T_a}{T_w}\right)^{-0.9}$$

or:

$$Nu = 1.10\left(\frac{4}{\pi}\,Gz\right)^{0.86}\left(\frac{T_a}{t_w}\right)^{-0.9}$$

Ceramic heat exchangers appear to hold much promise for use in temperature ranges above those generally encountered in conventional exchangers. However, no progress is in sight in their design and construction until methods are developed to construct mechanically reliable gastight seals, and means are found to prevent high thermal stresses. Both of these problems would arise in a commercial size heat exchanger of this type under operating conditions. Ceramic heat exchangers are also limited at present to low pressures that makes their use with a light, gaseous heat-transfer medium, such as helium, impractical.

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Nomenclature

A = heat transfer surface, ft.^{2}
C = specific heat at constant pressure, B.t.u./(lb.)(°F.)
D = inside diameter of tube or pipe, ft.
h = film heat-transfer coefficient, B.t.u./(hr.)(ft.^{2})(°F.)
k = thermal conductivity, B.t.u./(hr.)(ft.^{2}) (°F./ft.)
L = length, ft.
Q = heat flow, B.t.u./hr.
R = resistance to heat flow, (hr.)(ft.^{2})(°F.)/B.t.u.
T_i = temperature of helium at inlet, °F.
T_o = temperature of helium at outlet, °F.
T_a = average helium temperature, °F.
t_i = temperature of cooling gas at inlet, °F.
t_o = temperature of cooling gas at outlet, °F.
t_1, t_2, t_3 = wall temperatures at selected points along the tube length, °F.
t_w = average wall temperature, °F.
V = velocity, ft./hr.; or total volume, ft.^{3}
v = specific volume, ft.^{3}/lb.
W = weight flow rate of helium, lb./hr.
μ = viscosity, lb./(ft.)(hr.)
ρ = density, lb./ft.^{3}
x = distance, ft.
a, m, n, p, b = constants

Dimensionless Numbers

WC/kL = Graetz number (Gz)
hD/k = Nusselt number (Nu)
$C\mu/k$ = Prandtl number (Pr)
DG/μ = Reynolds number (Re)

Subscripts
i = inside of pipe or tube
o = outside of pipe or tube
f = film
a = hot gas
w = wall
m = mean value

Literature Cited

(1) Baroody, E. M., Duckworth, W. H., Simons, E. M., Schofield, H. Z.,"Effect of Shape and Material on the Thermal Rupture of Ceramics", U. S. Atomic Energy Comm., Nat'l Sci. Foundation, AECD - 3486, 5-57 (1951).
(2) Baroody, E. M., Simons, E. M., Duckworth, W. H., "Effect of Shape on Thermal Fracture", J. Am. Ceramic Soc., 38, (1955).
(3) Browning, S. C., Masters Thesis, West Virginia University, Morgantown, W. Va., 1959.
(4) Colburn, A. P., "Method of Correlating Forced Convection Heat Transfer Data and a Comparison with Fluid Friction", AIChE Trans., 29, 174 (1933).
(5) Deissler, R. G., "Investigation of Turbulent Flow and Heat Transfer in Smooth Tubes Including the Effects of Variable Properties", ASME Trans., 73, 101 (1951).
(6) Drew, T. B., "Mathematical Attacks on Forced Convection Problems: A Review", AIChE Trans., 26, 26 (1931).
(7) Drew, T. B., Hogan, J. J., McAdams, W. H., "Heat Transfer in Streamline Flow", Trans. AIChE, 26, 32 (1931).
(8) Groeber, H., Erk, S., Grigull, U., Die Grundgesetz der Warmeubertragung, Springer Verlag, Berlin, Germany, 3rd ed., 1955.
(9) Hilsenrath, J., Touloukian, Y. S., "The Viscosity, Thermal Conductivity and Prandtl Number for Air, O_2, N_2, NO, H_2, CO, CO_2, H_2O and A", ASME Trans., 79, 967-985 (1954).
(10) Humble, L. V., Lowdermilk, W. H., Desmond, L. G., "Measurements of Average Heat-Transfer and Friction Coefficients for Subsonic Flow of Air in Smooth Tubes at High Surface and Fluid Temperatures", NACA Report 1020, (1951).
(11) Knudsen, J. G., Katz, D. L., "Fluid Mechanics and Heat Transfer", McGraw -Hill, New York, 1958.
(12) Lee, J. F., Sears, F. W., "Thermodynamics", Addison Wesley, Reading, Mass., 1956.
(13) Lyon, R. N., "Heat Transfer at High Fluxes in Confined Spaces", Chem. Engr. Progr. 47, 75 (1951).
(14) Martinelli, R. C., "Heat Transfer to Molten Metals", ASME Trans., 69, 947 (1947).
(15) McAdams, W. H., "Heat Transmission", McGraw-Hill, New York, 1942.
(16) Norris, R. H., Streid, D. D., "Laminar-Flow Heat-Transfer Coefficients for Ducts", ASME Trans., 62, 525 (1940).
(17) Ramey, H. J., Henderson, J. B., Smith, J. M., "Heat-Transfer Coefficients for Gases; Effect of Temperature Level and Radiation", Chem. Eng. Progr. 50, Sym. Ser. No. 9, 21-28, (1954).
(18) Seban, R. A., Shimazaki, T. T., "Heat Transfer to a Fluid in a Smooth Pipe with Walls at a Constant Temperature", ASME Trans., 73, 803 (1951).
(19) Sieder, E. N., Tate, C. E., "Heat Transfer and Pressure Drop of Liquids in Tubes", Ind. Eng. Chem., 28, 1429 (1936).
(20) Zellnik, H. E., Churchill, S. W., "Convective Heat Transfer from High-Temperature Air Inside a Tube", Paper presented at ASME-AIChE Joint Heat Transfer Conference, (Aug. 1957).

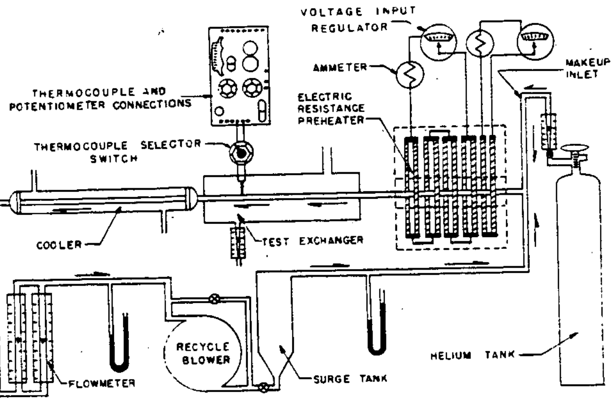

Figure 1. Flow diagram of apparatus for the study of heat transfer at high temperatures

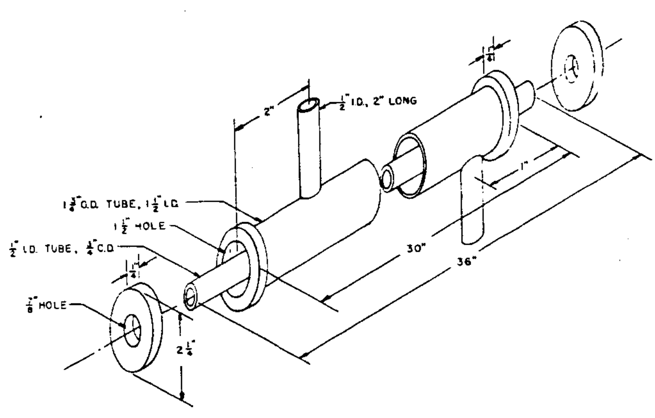

Figure 2. Construction details of vitreous alumina heat exchanger

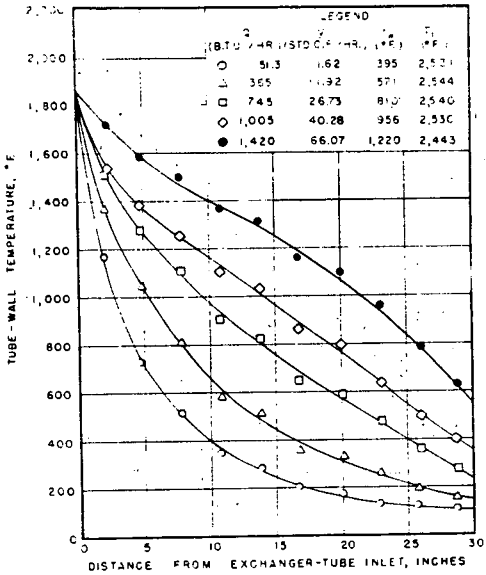

Figure 3. Distribution of tube-wall temperatures, counter flow

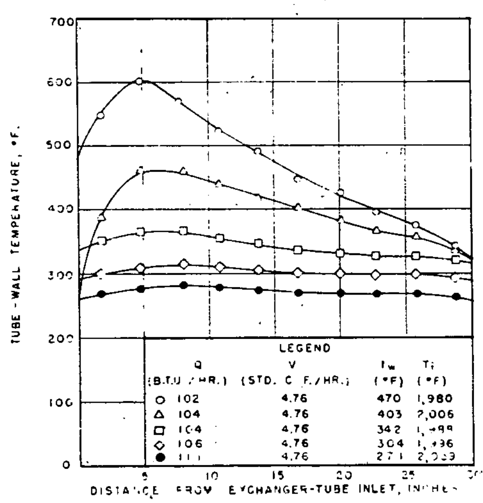

Figure 4. Distribution of tube-wall temperatures, parallel flow.

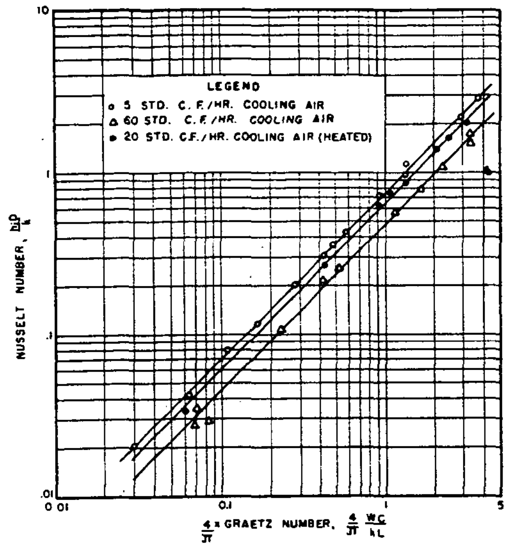

Figure 5. Correlation of heat-transfer data showing effect of cooling-air rate

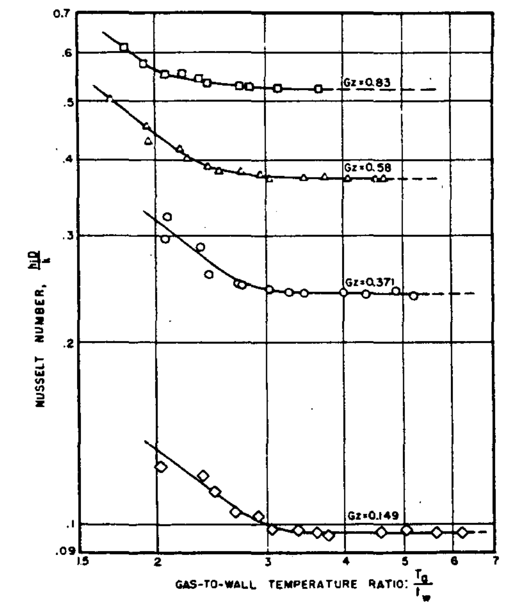

Figure 6. Effect of gas-to-wall temperature ratios on Nusselt number

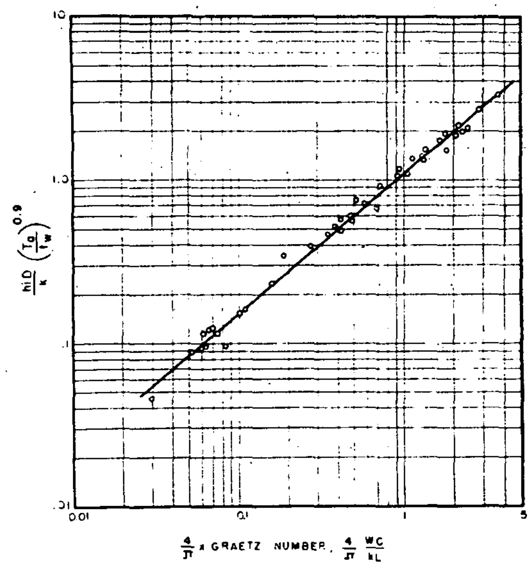

Figure 7. Correlation of heat-transfer data corrected for temperature ratios less than 3

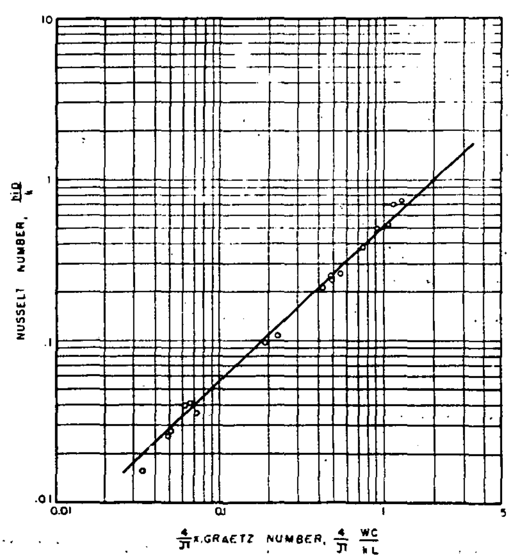

Figure 8. Correlation of heat-transfer data for temperature ratios greater than 3

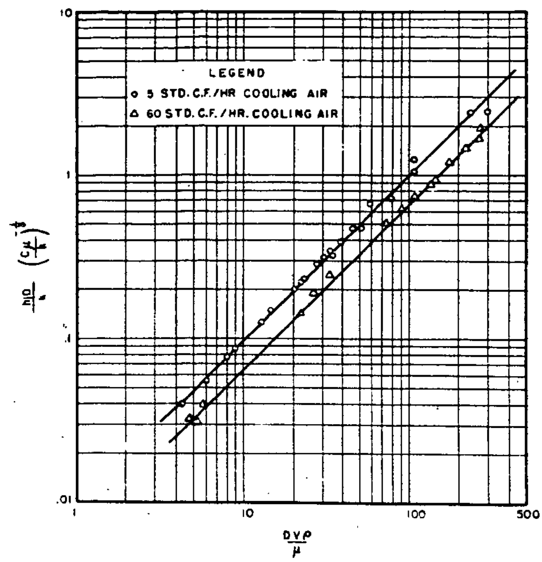

Figure 9. Correlation of heat-transfer data in terms of Reynolds number

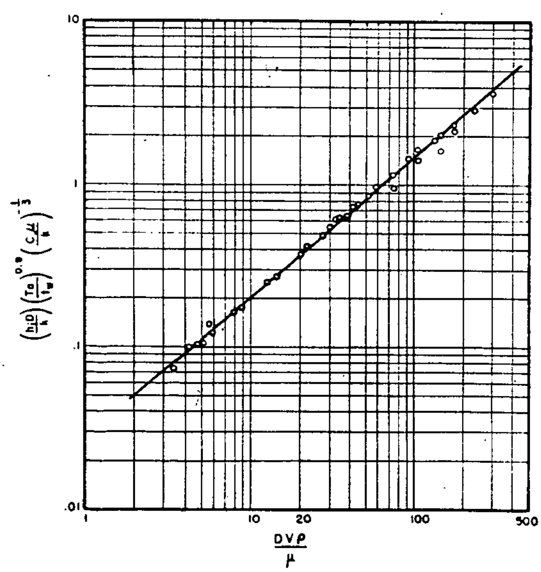

Figure 10. Correlation of heat-transfer data, corrected for temperature ratios, in terms of Reynolds number